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MODELING FLUID-STRUCTURE INTERACTION IN A BRAZILIAN GUITAR RESONANCE BOX

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Abstract. *An important aspect of musical acoustics research is to identify the relationship of measurable physical properties of a musical instrument with the subjective evaluation of their sound quality or tone. Therefore, it is important to develop analytical or numerical methods to predict accurately its vibroacoustic behavior. These methods will enable the determination of key parameters that can be used to control the tone and sound quality. This work uses theoretical modal analysis with finite element method (FEM) to determine the dynamic behavior of a Brazilian guitar resonance box in terms of natural frequency and mode shapes. In the sense of modeling the influence of compliance on the low frequency modes, the present work combines finite element model, which considers air cavity-resonance box coupling, with a Helmholtz resonator model, which considers the radiation effect through the sound hole by a neck length correction. In order to check the influence of this correction on the coupled modes, the obtained results are compared to a model which do not considers increment in the neck length. Additionally, experimental tests were made in the actual Brazilian guitar. The results are compared and discussed.*

Keywords: *vibroacoustic; Brazilian guitar; musical instruments; modal analysis.*

1. INTRODUCTION

The use of numerical models to determine the dynamic behavior of a musical instrument is a common way to study the relation between its physical properties and sound quality (Wright, 1997; Fletcher and Rossing, 2005; Hurtado *et al*, 2012). In other words, the determination of objective parameters may be used to obtain and control subjective requirements like performance and sound quality. In this sense, many works has developed methods for analytical and numerical prediction of the vibroacoustic behavior of string instruments With the advent of technology and consequent improvement of the computational processing, numerical models have been used to simulate complex systems and calculate modal parameters like vibration modes and natural frequencies, which are determinant in the tone and sound power desired for a musical instrument (Broke, 1992). Therefore, the use of these simulation tools seems to be valuable in the project of a musical instrument. By varying the structural and acoustic parameters it is possible to obtain different modal parameters without need for constructing multiple prototypes. Additionally, numerical methods for determining these characteristics in stringed instruments can help the makers improve the instrument design by avoiding empirical trial and error methods. For example, the *luthier*, craftsman specialized in the construction of musical instruments, do not have this freedom. In the building process of a musical instrument it is required to dealing with different types of wood, with well-defined characteristics, which makes impractical this type of investigation. Furthermore, the degree to which such features can be changed is limited not only by this interdependent relationship, but also by the need to finish the instrument, which must conform to aesthetic and mechanical requirements. Generally, guitars require a soundboard to resist the stresses imposed by string tension and need to have an attractive appearance in aesthetic standards.

This paper uses numerical modal analysis, calculated by finite element method (FEM), to determine the dynamic behavior of Brazilian guitar resonance box. The modal analysis technique allows to determine the natural frequencies and the corresponding mode shapes of structural, acoustic and vibroacoustic (acoustic and structural coupled) systems. It is known that both the air cavity and the structure of resonance box act as elastic elements in the coupled system and thus generate their corresponding vibration modes (Paiva, 2013). The cavity modes interact with the modes of the wood structure generating the vibration modes of the whole resonance box. In general, the Brazilian guitar have an internal cavity connected to the outside by one orifice and hence have his dynamic behavior greatly influenced by the environment in which is emerged. The lowest of the cavity modes, which do not consider coupled structure, is commonly called Helmholtz resonance, named as A0. The higher modes of the cavity (named as A1, A2, etc.) correspond to the stationary waves inside it and are not harmonically related to A0 (Elejabarrieta *et al*, 2002). These modes, specially the Helmholtz resonance, have been studied by several authors in the case of the guitar resonance box or the violin box and their influence on the instrument has been demonstrated (Jansson, 1977; Firth, 1977; Roberts, 1997).

Numerical simulations using the finite element method (FEM) have been previously applied to obtain the natural frequencies and the mode shapes of complex mechanical systems like musical instruments (Elejabarrieta *et al*, 2002a; Curtu *et al*, 2005). Paiva and Santos (2012) studied the Brazilian guitar resonance box and obtained the natural frequencies and corresponding mode shapes of acoustic (air cavity), structural (resonance box) and vibroacoustic (air

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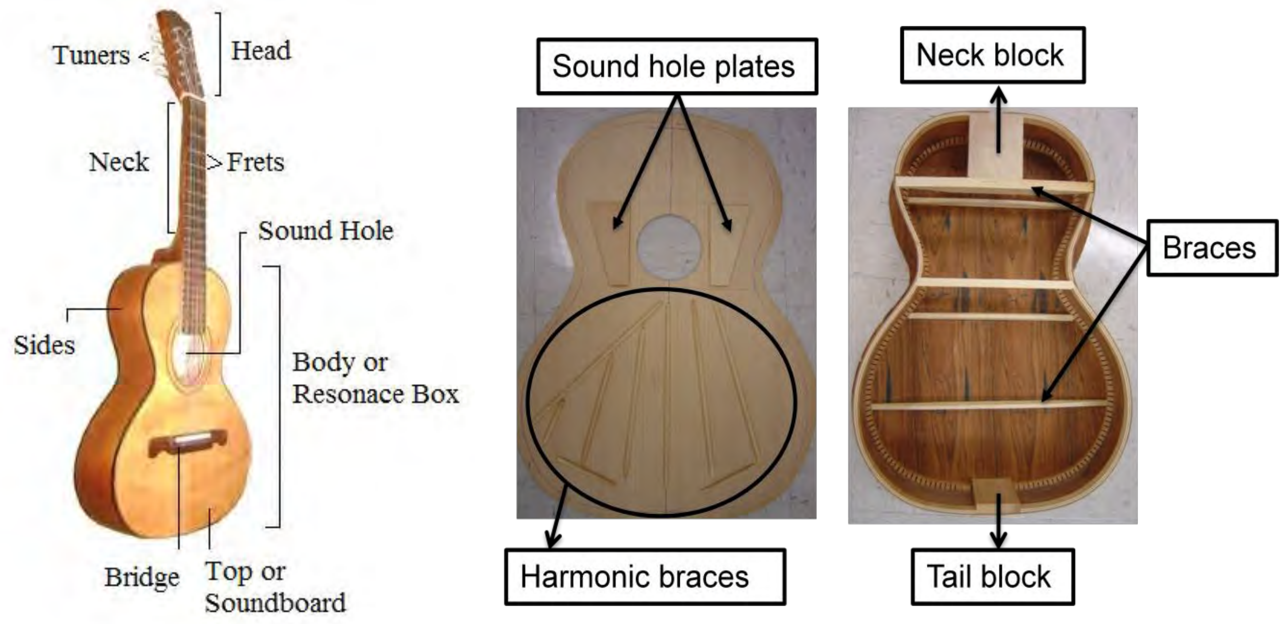
cavity-resonance box coupling) systems. In this work the air cavity was modeled with zero pressure boundary conditions at the location of the sound hole, with no mass loading or radiation impedance. This causes loss of accuracy for the Helmholtz resonance and for several of the higher modes. If the guitar cavity is approximated to a Helmholtz resonator, Kinsler *et al* (1982) suggest that the effect of radiation through the sound hole may be introduced by incrementing the length of the neck, which is called effective length. Here, this correction was applied to vibroacoustic system (air cavity-resonance box coupling) and the natural frequencies and mode shapes were compared to previous works. In order to validate the numerical simulations, the results of vibroacoustic model were updated.

2. THE BRAZILIAN GUITAR

The Brazilian guitar is a countryside musical instrument. It has different characteristics that vary regionally, configuring a sparse group of string musical instruments. The shape of resonance box, the arrangement of strings, the material properties and the tuning type are some of the characteristics that describe the instrument diversity. Thus, the expression "Brazilian Guitar" is able to qualify the instrument in all its variations. So many names and singularities for these guitars can be found along the Brazilian territory. This paper is focused on *viola Caipira*, which is the most known and played in all regions of Brazil, particularly in the Southeast and Midwest regions. Generally, the *viola Caipira* has 10 strings combined at five pairs. Two pairs are tuned in high notes on the same fundamental frequencies, i.e., the same note at the same frequency (unison), while the remaining pairs are tuned to the same note, but with a difference of one octave in their frequency (rate 2:1).

The *viola Caipira* is derived from the Portuguese guitar, which originates from Arabic instruments like lutes. In the fifteenth and sixteenth centuries, the Portuguese guitar was widespread in Portugal, being considered the main instrument of the minstrels and troubadours. It arrived in Brazil through the Portuguese settlers from different regions and has passed to be used by the Jesuits in the Indian catechesis (Vilela, 2004). Subsequently, the natives began to build rudimentarily these guitars using woods from Brazil. This instrument can be regarded as the forerunner of the *viola Caipira* we know today. At the beginning the *viola Caipira* practically kept the basic structure of its ancestor, following the same pattern.



(a) (b)

Figure 1. Main parts of a Brazilian guitar (*viola Caipira*): (*a*) External; and (*b*) Internal.

The main parts of a *viola Caipira* are similar to a classic guitar, as shown in Fig. 1. It is important that the strings are always adjusted to the proper tension, i.e., always respecting the structural capacity of the instrument, specially the soundboard and head, which are under constant compressive loading caused by the tension resulting in the tuners and bridge. There are also dynamic loads that are present while the instrument is played, and forces generated by gradients of temperature and humidity. Therefore, it is important to pay attention to the structural details when designing and building an instrument. The resonance box (or body) is composed by the top plate, back plate, sides and internal structures. These parts enclose the acoustic cavity, which communicates with the external air through the hole of the instrument (or sound hole). The strings are attached to the soundboard through the bridge as shown in Fig. 1a. It can be observed in Fig. 1b the internal fixings and reinforcements as: sound hole plates; harmonic braces; braces; lining, neck and tail blocks.

The dynamic behavior of a Brazilian guitar is determined by the interaction of many components that radiates sound in different ways. This is because sound behaves differently depending on the length of the sound wave compared to the dimension of the radiator. Rossing (1988) proposed the following scheme, as shown in Fig. 2, which presents two different ways for high and low frequencies sound.

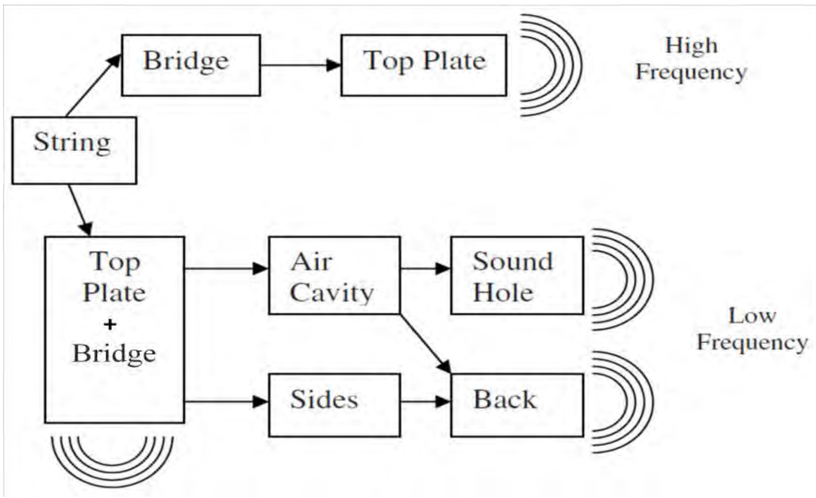


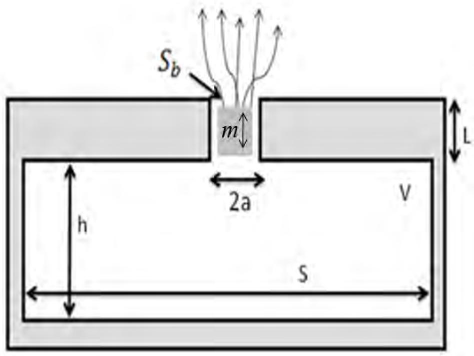
Figure 2. Schematic of radiation and energy flow in a guitar. (adapted from Rossing, 1988)

The production of sound starts in the interaction between the player's fingers and the strings. When a string is plucked, its vibration can be described in terms of transversal modes. The corresponding frequencies have an almost-harmonic relationship, and changing the plucked position the player excites different string modes and vary the tone quality. The strings radiate a little part of energy through the air, but in high frequencies the most energy is transmitted to soundboard via bridge. Otherwise, in low frequencies the bridge and the soundboard behave as unique structure, being excited directly by the strings.

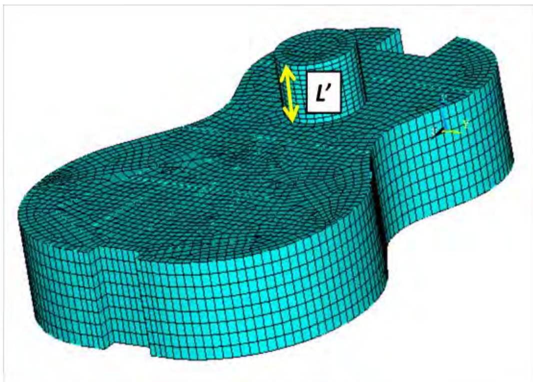
3. VIBROACOUSTIC MODAL ANALYSIS

3.1 Helmholtz resonator model

Analysis of many acoustic devices becomes simple if the wavelength in the fluid is much longer than the dimensions of the device. Acoustic devices in this long wavelength limit are termed *lumped acoustic devices* (Kinsler *et al*, 1982). A simple example of lumped acoustic system is the Helmholtz resonator. In order to simulate the effect of radiation in low frequencies through the sound hole, some works have modeled the cavities of string instruments by Helmholtz resonator approach (Elejabarrieta *et al*, 2002b; Bissinger, 2006; Cremer, 1984). Figure 3b shows one example applied to a finite element model of a Brazilian guitar, which will be highlighted later.



(a)



(b)

Figure 3. (a) Keys geometric parameters of a Helmholtz resonator. (b) Resulting mesh: combination of Helmholtz resonator model and finite element model.

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As shown in Fig. 3a, the resonator consists of a cavity of volume V (S is the area of base and h is the height), enclosed by rigid walls, filled with air connected to the outside by a neck of length L and cross-sectional area S_b. Therefore, the long wavelength limit implies $L << \lambda$, $S^{1/2} << \lambda$ and $V^{1/3} << \lambda$. Assuming that the inner and outer ends of the neck are equivalent to a flanged termination and the resonator walls are very thin compared to the dimensions of the hole, the following equations are valid (Kinsler *et al*, 1982):

$$m = \rho_0 S_b L' \tag{1}$$

$$L' = L + \Delta L \cong 1.6a \tag{2}$$

$$f_{A0} = (c_0/2\pi)\sqrt{S_b/V(L+\Delta L)} \tag{3}$$

where m is the total effective mass around the sound hole, ρ_0 is the density of the fluid, c_0 is the sound velocity of in the fluid, f_{A0} is the frequency of the Helmholtz resonator, L' is the effective length of the neck and ΔL represents the contribution of the fluid through the sound hole. This correction depends on the shape and size of the orifice, and adopts different values depending on these parameters. In this model, the frequency of the resonator does not depend on the shape of the cavity but only on its volume and the dimensions of the orifice. Moreover as has been pointed out in reference (Dickens, 1978), the proximity of the back and ribs to the opening has an additional effect in the case of the guitar. So the analytical calculation of the length correction ΔL may take into account the geometrical details of the cavity.

Bissinger (2006) discussed a necessary correction in the Helmholtz resonator frequency due the compliance of the walls, which must be considered as elastics. This compliance is caused by the pressure exerted on the walls, increases with instrument volume and, generally, reduces the cavity mode frequencies. Therefore, in the sense of modeling the influence of compliance on the low frequency modes, the present work combines finite element model, which considers air cavity-resonance box coupling, with a Helmholtz resonator model, which considers the radiation effect through the sound hole. According to Eq.'s (1), (2) and (3), the parameters used to perform this model, which uses dimensions of an actual Brazilian guitar detailed in section 3.3, are shown in Tab. 1.

Table 1. Parameters of Helmholtz resonator according Eq.'s (1), (2) and (3).

ρ_0 [Kg/m^3]	m [Kg]	S_b.[m^2]	L' [m]	L [m]	a [m]	c_0 [m/s]	V [m^3]	f_{A0} [Hz]
1.125	2.0x10^{-4}	3.8x10^{-3}	4.8x10^{-2}	3.0x10^{-3}	3.0x10^{-2}	343	0.85x10^{-2}	165.0

3.2 Finite element method: theorical basis

This section presents a concise mathematical formulation of the finite element method for the vibroacoustic (coupling between solid and fluid domains) modal analysis. In the coupled field it is necessary to take into account the interaction effects between the fields at the interface solid-fluid. The formulation presented here was developed by Zienkiewicz and Newton (1969). Based on the theory of elasticity, the dynamic behavior of a linear elastic solid (small deformations) can be written in index notation as:

$$\sigma_{ij,j} + f_i = \rho_s \ddot{u}_i \tag{4}$$

where σ_{ij} is the stress tensor, f_i is the vector of body forces, ρ_s is the solid density, u_i is the displacement, and i and $j = x$, y, z. The effect of fluid over the solid is included in the interfaces trough the fluid pressure over the solid surface, i.e., the balance of forces in the normal direction to the field interfaces must be imposed as

$$\sigma_{ij}\, n_i = n_i p \tag{5}$$

where n_i is the interface normal vector and p is the acoustic pressure. The acoustic wave equation can be written as:

$$\nabla^2 p + \frac{1}{c^2}\ddot{p} = -g \tag{6}$$

where c is the sound speed and g is the source field. The effect of the solid over the fluid is also considered in the domain interfaces through the kinematic compatibility of the solid in contact with the fluid, namely

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$$\frac{\partial p}{\partial n} = -\rho_f \ddot{u}_n \tag{7}$$

where ρ_f is the fluid density and u_n is the displacement component normal to the interface. Applying the weighted residual method and discretizing with finite elements on Eqs. (4) to (7), we obtain the element mass matrix, element stiffness matrix, element volumetric stiffness matrix, element compressibility matrix and element interface matrix as:

$$\mathbf{M}^e = \int_\Omega \rho_s \mathbf{N}_s^T \mathbf{N}_s \, d\Omega \, ; \; \mathbf{K}^e = \int_\Omega \mathbf{B}_s^T \mathbf{D} \mathbf{B}_s \, d\Omega \, ; \; \mathbf{E}^e = \frac{1}{c^2} \int_\Psi \rho_f \mathbf{N}_f^T \mathbf{N}_f \, d\Psi \, ; \; \mathbf{H}^e = \int_\Psi \mathbf{B}_f^T \mathbf{B}_f \, d\Psi \, ; \; \mathbf{L}^{e^T} = \int_{\Gamma_i} \mathbf{N}_f^T \mathbf{n} \mathbf{N}_s \, d\Gamma_i \tag{8}$$

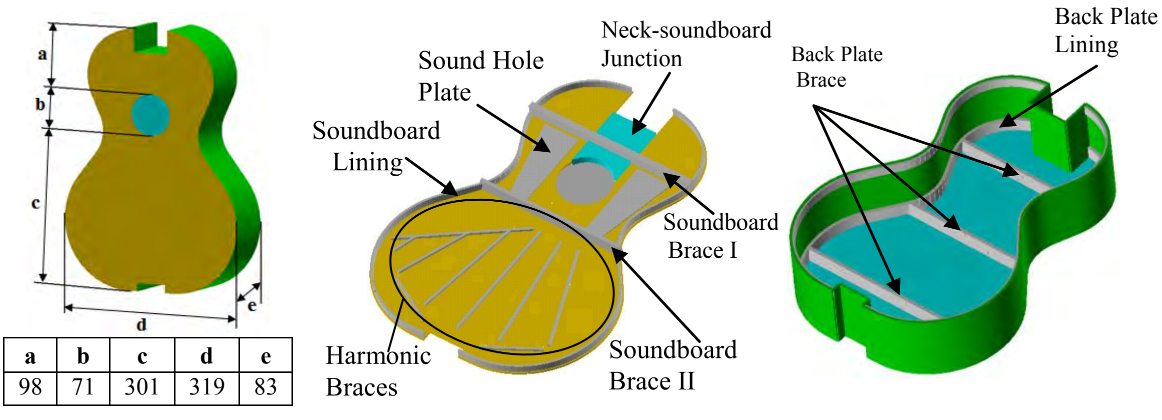
where **N** is the element shape function matrix, **B** is the nodal strain-displacement matrix, **D** is the constitutive law matrix and the index $_s$ and $_f$ refers to solid and fluid domain, respectively. The Greek letters Ω, Ψ and Γ refers to the geometry of the structural, fluid and interface domain, respectively. Writing the equations to the coupled system in terms of a global matrix in the frequency domain and considering the free vibration condition we have:

$$\left(\begin{bmatrix} \mathbf{K} & -\mathbf{L} \\ \mathbf{0} & \mathbf{H} \end{bmatrix} - \boldsymbol{\Lambda}_a \begin{bmatrix} \mathbf{M} & \mathbf{0} \\ \rho_f \mathbf{L}^T & \mathbf{E} \end{bmatrix} \right) \begin{Bmatrix} \mathbf{d} \\ \mathbf{p} \end{Bmatrix} = \begin{Bmatrix} \mathbf{0} \\ \mathbf{0} \end{Bmatrix} \tag{9}$$

where Λ_a is a diagonal matrix of the square of the natural frequency of coupled domain and $\{\mathbf{d}\ \mathbf{p}\}^T$ is the displacement-acoustic pressure nodal vectors of the corresponding vibration mode shapes. This solution is also known as *vibroacoustic modal analysis*.

3.3 Resonance box numerical model

The finite element computer model geometry of the resonance box was built in the software ANSYS 13.0, using the parts and dimensions of a commercial *viola Caipira*, brand *Rozini*, *Ponteio Profissional* model. Figure 4 shows the guitar main dimensions and the components included in the Finite Element (FE) model.



a	b	c	d	e
98	71	301	319	83

(*a*) Dimension in mm. (*b*) Soundboard (*c*) Back Plate and Sides

Figure 4. Rozini Brazilian guitar resonance box: (*a*) Main dimensions. (b) (c) Components included in FE model.

The difficulty to identify the wood of resonance box components led to the choices by indications from the literature (Bergman *et al*, 2010). Furthermore, when it was possible to identify the wood component, was not found its mechanical properties, leading to the use of a similar timber properties. Thus, it is considered that the soundboard and internal reinforcements are made from Sitka Spruce, and the back plate, sides and neck-soundboard junction are made of Yellow Birch (*Betula alleghaniensis*). Table 2 shows the mechanical properties of woods used in FE model. The thicknesses of soundboard, back plate and sides are 3.0 mm, 3.5 mm and 2.0 mm, respectively. Table 3 shows the cross section dimensions of braces and linings included in FE model.

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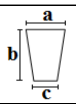
Table 2. Wood mechanical properties (Bergman *et al*, 2010)

Wood	E_x [MPa]	E_y [MPa]	E_z [MPa]	G_{xz} [MPa]	G_{xy} [MPa]	G_{yz} [MPa]	ν_{xz}	ν_{xy}	ν_{zy}	ρ [Kg/m^3]
Spruce	10.340	800	440	660	630	30	0.372	0.467	0.435	460
Birch	11.320	880	560	830	760	190	0.426	0.451	0.697	668

Table 3. Cross section dimensions of braces and linings.

Cross Section	**Dimensions [mm]**	**Harmonic Brace**	**Soundboard Brace I**	**Soundboard Brace II**	**Back Plate Brace**	**Soundboard Lining**	**Back Plate Lining**
	**a**	3.5	8.0	9.0	9.0	3.0	4.5
	b	3.5	18.0	5.0	5.0	12.0	16.0
	c	3.5	8.0	16.0	16.0	3.0	4.5

The vibroacoustic model considers the resonance box structure filled with air, where the air is modeled with FLUID30 element, while the structure is modeled as SHELL63 element for plates and with BEAM188 element for beams. Fluid-structure interaction is obtained with the coupling matrices, which lead to a solution of an eigenproblem with asymmetric matrices. The ANSYS imposes this condition through the FSI command applied in fluid-structure contact surface. The resonator model is implemented, as show in section 3.1. The neck length is therefore increased which provides the correction proposed by Eq. (3) and the definitive mesh built up, as shown in Fig. 3b, setting the sound pressure to zero on top of the neck correction. This numeric model also includes the orthotropic wood properties and is constructed in solid and fluid geometries, which contains a total number of 23,429 elements and 20,109 nodes. A vibroacoustic modal analysis is performed, where the free boundary condition was applied to the structure. In order to check the influence of Helmholtz resonator correction on the coupled modes, Tab. 4 shows comparison between the first 5 natural frequencies and mode shapes of two different models: one will be called HR model (Helmholtz resonator model), which considers increment in the neck length, and the other will be called simple model, which does not.

In the structural displacements results of Tab.4 and Tab.6 the blue color indicates the nodal regions, i.e., without displacement, and red regions represents maximum amplitude of displacement. In the acoustic pressure results of Tab. 4 the blue and red colors indicate maximum amplitude of acoustic pressure, being differentiated by opposite phases; green color indicate null pressure zones. Table 4 explains the influence of neck increment on the resonance box modes. It is evident that the structural displacement distribution was not affected because their mode shapes remain the same. The great difference was noted in the distribution of pressure acoustics. As expected, the largest differences were obtained for the first three modes, which have significantly changed their mode shape. The fourth and fifth mode shapes changed partially, with a slight difference in the pressure distribution. The values of natural frequencies were influenced in the same direction, i.e., for the first three modes the changes were significant, producing differences of 34.3, 5.5 and 4.5%, respectively. For the fourth and fifth modes changes were insignificant, presenting, as shown in Tab. 4, null differences.

It is clear that the first mode is not a pure Helmholtz mode because the sound pressure in the cavity does not feature a constant distribution. This result is expected for HR model since the modeled resonance box is not an ideal Helmholtz resonator and its walls are not rigid. Moreover, the sound pressure in the zone of the sound hole varies perpendicularly. Although the results show an in-phase vibrational pattern, they also reveal a different value for the sound pressure between the upper and lower part of the cavity.

To verify this numerical results, experimental test are made in the actual *viola*. Experimental test was conducted to the 5 first natural frequencies and mode shapes. Test was performed in an anechoic chamber in a complete *viola* (no strings), instead of the resonance box as in numerical model. By using a sine sweep signal, an acoustic excitation (loudspeaker) is applied to the guitar suspended by elastic bands (free condition). The natural frequencies are obtained by determining the resonance peaks of the soundboard speed (laser vibrometer) or acoustic pressure (microphone) inside the resonance box. Then mode shapes of the structure are obtained using the figures of Chladni. To obtain these figures the guitar is excited in each natural frequency, and spreading minced leaves of tea on the soundboard these will accumulate on the regions of nodal lines (zero speed) revealing the mode shape. Figure 5 shows the arrangement of experimental setup and a Chladni figure.

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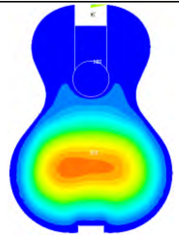
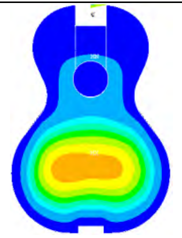
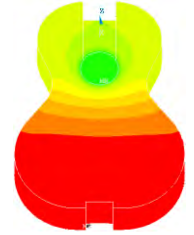
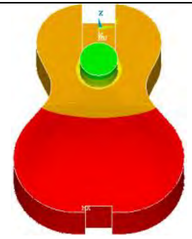
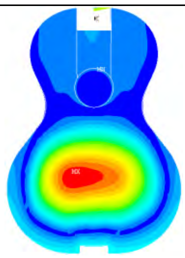
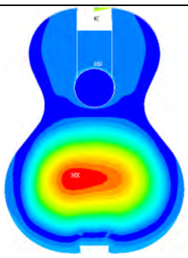
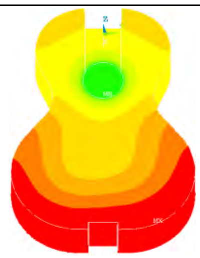
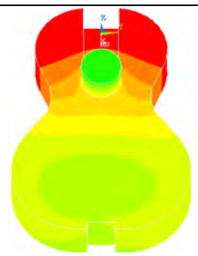
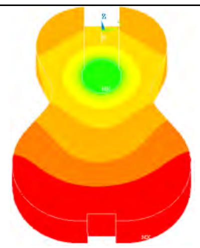
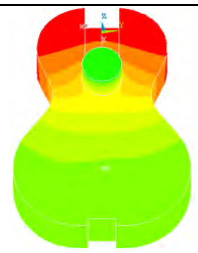
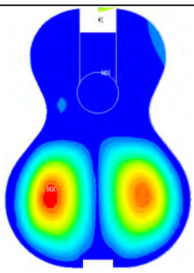
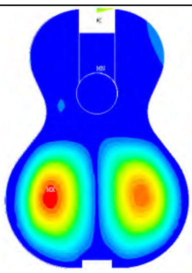
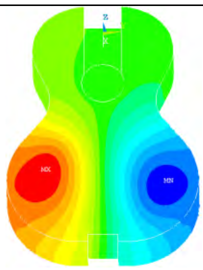
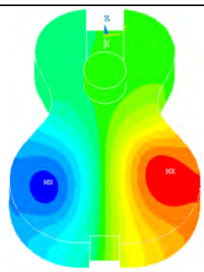
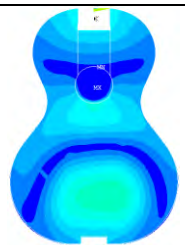
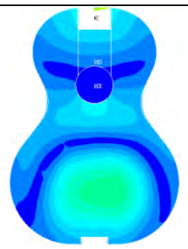
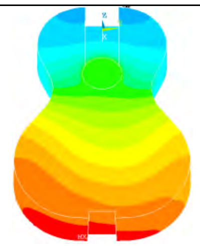
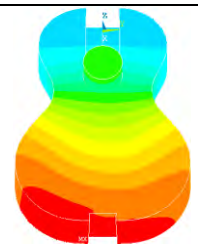
Table 4. First 5 vibroacoustic natural frequencies and mode shapes of the guitar resonance box to Simple and HR models.

No.	STRUCTURAL DISPLACEMENT		ACOUSTIC PRESSURE		DIFFERENCE [%]
	SIMPLE MODEL	RH MODEL	SIMPLE MODEL	RH MODEL	
Mode 1	 169 Hz	 111 Hz	 169 Hz	 111 Hz	34.3
Mode 2	 309 Hz	 295 Hz	 309 Hz	 295 Hz	4.5
Mode 3	 341 Hz	 322 Hz	 341 Hz	 322 Hz	5.5
Mode 4	 361 Hz	 361 Hz	 361 Hz	 361 Hz	0.0
Mode 5	 384 Hz	 381 Hz	 384 Hz	 381 Hz	~0

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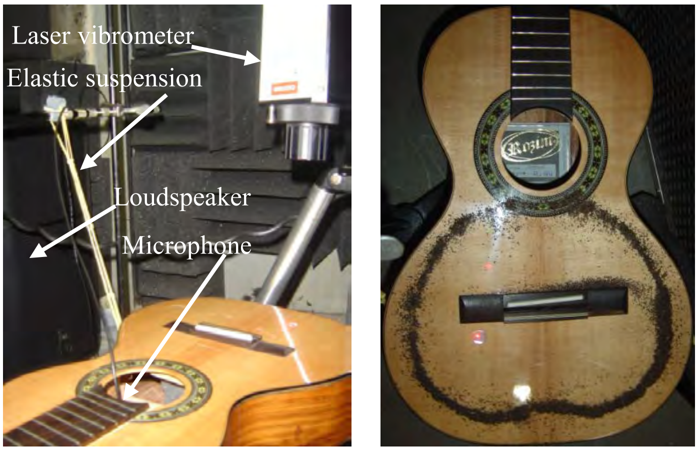

(*a*) (*b*)

Figure 5 – Brazilian guitar: (a) experimental setup; and (b) Chladni figure.

Table. 6 presents the first 5 structural natural frequencies and mode shapes obtained with the numerical HR model and the experimental measurements on the actual *viola*. Table 6 presents the relative error between the numerical models, analytical equation, Eq. (3), and experimental resonance frequencies.

The technique of Chladni figures reveals clearly only the experimental modes with enough power to push the particles of tea to the nodal lines. For the *viola* test only the two first modes let us to observe the experimental mode shapes in soundboard and back plate. The numerical modes 1 and 2 of the soundboard and the modes 1 and 2 of back plate are in good agreement with the corresponding results of experimental modes. For the experimental modes 3, 4 and 5 were identified only one mode shape. The experimental modes 3 and 5 occur on back plate and present good agreement with the numerical result. The experimental mode 4 occurs on soundboard and do not have conflicting forms with their numerical modes. From the results presented it is observed that there are modes that have not found their shapes due to limitations of the experimental technique. This assumes great contrasts between regions of significant displacement and regions of zero displacement. Moreover, the numerical mode shapes are more satisfactory than the results obtained previously (Paiva and Santos, 2012), which do not consider Helmholtz resonator correction. In general, the comparative results in Tab. 5 confirm the validity of HR model taking account the smaller errors for natural frequency values that were obtained.

The disagreement between some results of natural frequencies and mode shapes can also be attributed to the simplifications assumed in computer simulation. Some of the wood components of the resonance box had their mechanical properties substituted by that of similar wood. Also, the lacquer layer, the presence of the neck and the influence of external fluid were not considered in the computational models.

Therefore, the comparative results in Tab. 5 confirm the validity of HR model. It is clear that the smaller errors for natural frequency values were obtained for this model. Additionally, is observed that both Simple and HR models present smaller errors for the highest modes.

The disagreement between some results of natural frequencies and mode shapes can also be attributed to the simplifications assumed in computer simulation. Some of the wood components of the resonance box had their mechanical properties substituted by that of similar wood. Also, the lacquer layer, the presence of the neck and the influence of external fluid were not considered in the computational models.

Table 5. Comparison of the first 5 natural frequencies obtained by the experiment and the numerical models.

	Experimental	**Simple model (only FEM)**		**HR model (FEM+HR)**		**Eq.(3)** (Kinsler *et al*, 1982)	
Mode	Frequency(Hz)	Frequency(Hz)	Error (%)	Frequency(Hz)	Error (%)	Frequency(Hz)	Error (%)
1	135	169	25.2	111	17.5	165*	22.2
2	266	309	13.9	295	9.8	----	----
3	321	341	5.8	322	0.3	----	----
4	340	361	5.8	361	5.8	----	----
5	374	384	2.6	381	1.8	----	----

* Equation (3) considers the air cavity enclosed by rigid walls and takes into account only the Helmholtz resonance frequency, f_{A0}.

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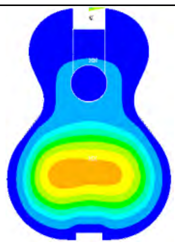
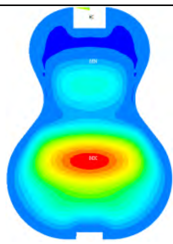
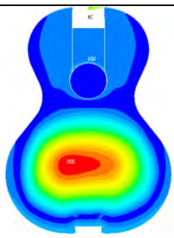
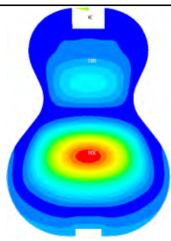
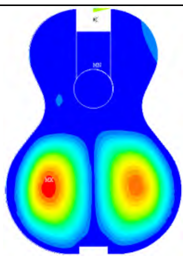
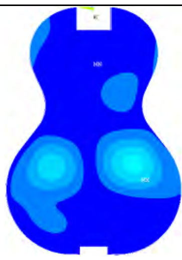
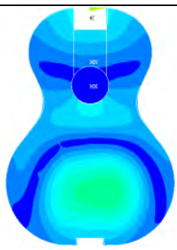
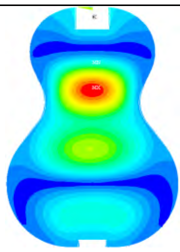
Table 6. First 5 numerical (HR model) and experimental natural frequencies and mode shapes of the guitar resonance box.

No.	SOUNDBOARD		BACK PLATE	
	Numerical	Experimental	Numerical	Experimental
Mode 1	 111 Hz	 135 Hz	 111 Hz	 135 Hz
Mode 2	 295 Hz	 266 Hz	 295 Hz	 266 Hz
Mode 3	 322 Hz	**N/A**	 322 Hz	 321 Hz
Mode 4	 361 Hz	 340 Hz	 361 Hz	**N/A**
Mode 5	 381 Hz	**N/A**	 381 Hz	 374 Hz

4. CONCLUSION

The Brazilian guitar was briefly described with regard to its historical, structural and acoustical characteristics. A procedure for vibroacoustic modal analysis by finite element method with ANSYS software was developed. It is clear the influence of neck length correction (Helmholtz resonator model) on the numerical results. The largest differences were noted in the distribution of acoustic pressure, specially in the first three modes. The fourth and fifth mode changed partially, with a small difference in their shapes. The values of natural frequencies were influenced in the same direction, i.e., for the first three modes the changes were significant. For the fourth and fifth modes changes were null.

Additionally, to validate the numerical results, experimental test was made in the actual guitar using Chladni technique. In general, it was observed that the finite element method proved to be effective and can determine the dynamic behavior of the resonance box of the guitar. However, in order to get better results from the computational model it is very important to indentify all woods used and its mechanical properties. It was shown that the experimental technique of Chladni figures has certain limitations to identify modes with low energy content. This suggests for a next stage of the work, carrying out an experimental modal analysis, a method which is more complete and efficient. Furthermore, it must be observed that the determination of the modal characteristics of the resonance box is only an initial stage about the study to obtain tones and sound power desired for stringed instruments. But, this information is already a helpful tool for *luthiers* and manufactures to get more control over the quality of the instrument in different stages of its construction. Finally, to go further in this research we need additional tests and psychoacoustic measurements of the sound field, which explain subjective evaluations and allow their relationship with objective parameters.

5. ACKNOWLEDGEMENTS

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